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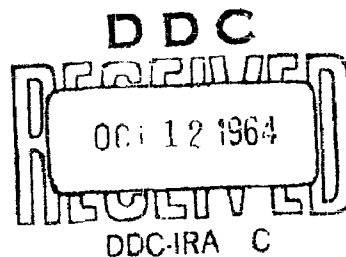
AIR-FILM COOLING OF A SUPERSONIC  
NOZZLE

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AIR-FILM COOLING OF A SUPERSONIC NOZZLE

by  
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**ABSTRACT:** An experimental study was made of the internal air-film cooling of a Mach 2.4, nonadiabatic wall, axially symmetric nozzle. The main stream air was heated to supply temperatures from 672 to 1212°R at supply pressures from 115 to 465 psia. The film coolant air was injected through a single peripheral slot at an angle of 10° from the nozzle wall. The coolant-to-main stream mass flow ratios were varied up to 20 percent. Steady-state nozzle wall temperatures were measured in both the subsonic and the supersonic flow regimes.

The turbulent pipe flow equation of Dittus and Boelter was found to be applicable in predicting the heat transfer rates in the absence of film cooling. A modified version of the semi-empirical equation of Hatch and Papell was found applicable in estimating the film-cooled nozzle wall temperatures.

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WHITE OAK, MARYLAND

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Air-Film Cooling of a Supersonic Nozzle

This report presents a comparison between experimental data and theoretical calculations of air-film cooling of an axisymmetric, Mach 2.4 nozzle.

A number of people have contributed towards this project. Special thanks are due to Mr. R. C. Sullivan for the instrumentation of the nozzle and Mr. F. W. Brown for his general assistance in performing the experiment.

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By direction

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## SYMBOL LIST

(The units listed here are consistent with the equations given in this report.)

A	flow area, $\text{in}^2$
$c_p$	specific heat at constant pressure, $\text{Btu-in}^2/\text{lbm-ft}^2\text{-}^\circ\text{R}$
d	diameter, in
f	velocity ratio function defined by equations (21) and (22)
h	heat transfer coefficient, $\text{Btu/sec-ft}^2\text{-}^\circ\text{R}$
k	thermal conductivity, $\text{Btu-in/sec-ft}^2\text{-}^\circ\text{R}$
L	injection slot length = $2\pi r$ for axisymmetric slot, in
M	Mach number
$\dot{m}$	mass flow rate, $\text{lbm/sec}$
Nu	Nusselt number
P	pressure, $\text{lbf/in}^2$ abs.
p	pressure ratio defined by equation (3)
Pr	Prandtl number
q	heat flux, $\text{Btu/sec-ft}^2$
r	radius, in
Re	Reynolds number
S	injection slot width, in
T	temperature, $^\circ\text{R}$
u	velocity, $\text{ft/sec}$
z	axial distance measured from the injection slot exit, in
$\alpha$	thermal diffusivity, $\text{in-ft/sec}$
$\beta$	effective injection angle defined by equation (19), radians

$\epsilon$	geometrical injection angle, radians
$\eta$	adiabatic wall film-cooling effectiveness
$\eta'$	nonadiabatic wall film-cooling effectiveness
$\mu$	absolute viscosity, lbm/sec-in
$\rho$	mass density, lbm/ft-in <sup>3</sup>
$\phi$	exponent defined by equation (20)
$\psi$	mass flow parameter defined by equation (4). lbf/in <sup>2</sup> abs/OR <sup>1/2</sup>

### Subscripts

a	upstream of orifice in mass flow meter
aw	adiabatic wall
b	downstream of orifice in mass flow meter
c	coolant, at injection slot exit
d	local diameter
g	main stream
H	cooling water
i	nozzle inner surface
j	inner surface thermocouple junction locations
o	nozzle outer surface
s	main stream supply (stagnation conditions)
w	nozzle wall
$\infty$	local free stream

### Superscripts

o	zero coolant injection
*	conditions at throat

## INTRODUCTION

When a surface is exposed to a high energy fluid stream, some means of cooling the surface may become necessary. Many methods of cooling are available or have been proposed (ref. (1)). Among these is film cooling. Film cooling employs a second fluid (the coolant) introduced between the surface and the high energy main stream, thus absorbing the heat which would otherwise flow from the main stream to the surface. The coolant, not necessarily the same fluid as the main stream, can be injected through a single slot or a series of slots in the surface.

In any particular application, many factors need to be considered in selecting a cooling method. Some of the features of film cooling are the following:

- (1) Film cooling may be applied when needed.
- (2) Film cooling may be applied locally (single-slot arrangement) or over an extended surface (multiple-slot technique (ref. (2))).
- (3) Film cooling does not alter the geometry of the surface (as in ablative cooling).
- (4) Film cooling does not require a special material or manufacturing process (as in porous-wall transpiration cooling).

The application of film cooling has been studied quite extensively (e.g., refs. (1) through (8)). The simplified flow model of Hatch and Papell (ref. (6)), developed for the gaseous film cooling of an adiabatic plate, resulted in a semi-empirical equation. This equation, incorporating the injection slot angle factor of reference (7), was used with some success by Lucas and Golladay (ref. (8)) in correlating their experimental data for film cooling of a rocket motor.

In the present investigation, the air-film cooling of an axially symmetric, nonadiabatic wall, Mach 2.4, contoured nozzle was studied by relating the decrease in the local nozzle wall temperatures with the increase in coolant mass flow rate. The equation of reference (7) was modified to apply to the axisymmetric, nonadiabatic wall used in the experiment.

## EXPERIMENTAL PROCEDURE

Apparatus and Instrumentation

The air-film cooling of a supersonic nozzle was performed with the apparatus shown schematically in figure 1. The compressed air in the bottle field was used as the supply for both the main stream and the film coolant. The main stream air was heated in a propane-fired, indirect heat exchanger, the supply conditions being measured with a total temperature thermocouple and a static pressure tap, both located in a plenum chamber to be described below. The coolant air was tapped directly from the bottle field via a 1-inch line and metered with an orifice of 0.101-inch diameter. The coolant mass flow was controlled by a valve upstream of the orifice.

The essential features of the film cooling apparatus (fig. 1) consisted of a plenum chamber, a contraction section, an approach section, a coolant injection slot, and a nozzle. The plenum chamber is 54 inches long by  $5\frac{1}{2}$  inches ID. The contraction section is contoured,  $8\frac{1}{2}$  inches long by  $5\frac{1}{2}$ -inch/1-inch ID. The approach section is 6 inches long by 1 inch ID. The nozzle (fig. 2a) is 13 inches long, is made of a chromium-copper alloy, and is externally water-cooled. The inlet and the exit sections of the nozzle are both 1 inch in diameter; the throat is 0.645-inch in diameter and 5 inches downstream of the injection slot exit. The design flow characteristics along the nozzle centerline are shown in figure 2b. The coolant injection slot (fig. 2c) has a width,  $S$ , of 0.0087-inch and a discharge angle,  $\epsilon$ , of  $10^\circ$  measured from the nozzle wall.

Fifteen pairs of chromel-alumel thermocouples embedded in the nozzle wall at fifteen axial locations measured the temperature distributions along the inner and outer surfaces. These thermocouples were embedded approximately 0.03 inches from the inner wall and mounted on the outer wall. The fifteen thermocouple stations covered a range  $1.25'' \leq z \leq 9.00''$ , as shown in figure 2. These thermocouples were displaced circumferentially along the nozzle for ease of installation and reduction of stress. An axisymmetric temperature field in the nozzle wall is assumed.

The thermocouples measuring the nozzle inner surface temperatures were cemented in tapered plugs made of the same material as the nozzle such that the junctions protrude from the side of the plugs (fig. 4). These plugs were force-fitted and cemented into mating, tapered holes in the nozzle wall. The nozzle inner surface was machined after the plugs were installed, thus ensuring a smooth surface. The thermocouples measuring the outer surface temperatures were peened and cemented into small

indentations on the surface. This technique of thermocouple installation was used so as not to perturb the temperature field in the nozzle wall. The nozzle and thermocouple assembly was inserted and sealed inside a water jacket for the nonadiabatic test conditions (fig. 2a).

The thermocouple outputs were referenced to "cold" junctions in an oil bath at ambient temperature. The oil bath temperature was measured with another Cr-Al thermocouple with its cold junction in an ice bath. This thermocouple read-out method, shown schematically in figure 5, required only one ice bath and relatively short thermocouple wires. The validity of the method is insured by two fundamental laws of thermoelectric thermometry; viz., the law of intermediate metals and the law of successive temperatures (ref. (9)).

The output of all the thermocouples was recorded on magnetic tapes by an electronic digital read-out system.

#### Test Conditions and Procedure

The experimental conditions used are given in the following table:

Main Stream Supply Conditions		Main Stream Mass Flow Rate Without Film Cooling	Film Coolant Mass Flow Rates	
$T_s$ (°R)	$P_s$ (psia)	$\dot{m}_g$ (lbm/sec)	$(\dot{m}_c)_{min}$ (lbm/sec)	$(\dot{m}_c)_{max}$ (lbm/sec)
672	315	2.113	0.0737	0.2088
852	315	1.876	0.0409	0.2085
852	465	2.770	0.0605	0.2020
1032	115	0.6224	0.0368	0.0778
1032	215	1.164	0.0453	0.2324
1032	315	1.705	0.0742	0.2102
1032	465	2.517	(no film cooling data)	
1212	115	0.5743	0.0390	0.0953
1212	315	1.573	0.0739	0.2097

The main stream supply conditions were maintained constant for each test ( $\pm 4^\circ\text{R}$  and  $\pm 1.5$  psi) by an automatic control system. The water flow rate was maintained at a constant value high enough to prevent local boiling.

The experimental procedure was as follows: For each of the supply conditions used, the nozzle wall temperatures without film cooling were recorded after they had reached steady-state values. Different amounts of film coolant were then injected into the nozzle and the steady-state wall temperatures were recorded for each coolant flow rate.

### ANALYSES AND RESULTS

The experimental data were reduced and analyzed to yield information concerning nozzle heat transfer without film cooling and nozzle wall temperature reduction with film cooling.

The basic quantities and parameters needed for the analyses were determined by the following relations and assumptions:

#### (1) Mass Flow Rates

- (a) The main stream mass flow rate is (assuming one-dimensional isentropic flow, sonic throat,  $d^*=0.645"$ , and no film coolant)

$$\dot{m}_g^0 = 0.174 P_s / \sqrt{T_s} \quad (1)$$

- (b) The film coolant mass flow rate is, (from the standard ASME orifice flow equations for air (ref. (10))) for subcritical orifice flow ( $p \geq 0.535$ ),

$$\dot{m}_c = 5.19 \times 10^{-3} \psi \sqrt{(1-p)(1.0755p - 0.0755)} \quad (2)$$

where

$$p = P_b / P_a \quad (3)$$

$$\psi = (0.707 + 0.293p) P_a / \sqrt{T_a} \quad (4)$$

and for critical orifice flow ( $p \leq 0.535$ ),

$$\dot{m}_c = 2.51 \times 10^{-3} \psi \quad (5)$$

(2) The thermocouples measuring the inner surface temperatures were approximately 0.03 inch from the inner surface (fig. 3). These thermocouple readings were assumed to correspond to the inner surface temperatures (i.e.,  $T_{wj} = T_{wi}$ ) in determining the temperature difference in equation (10) and in computing the film cooling effectiveness defined by equation (23).

### (3) Properties of Main Stream Air

(a) The absolute viscosity is (from ref. (11))

$$\mu = \frac{0.609 \times 10^{-4} (T/100)^{3/2}}{198.7 + T}, \text{ lbm/sec-in} \quad (6)$$

(b) The thermal conductivity is (from ref. (11))

$$k = \frac{0.38 \times 10^{-2} (T/100)^{3/2}}{441.7 \times 10^{-2} 1.6/T + T}, \text{ Btu-in/sec-ft}^2\text{-R} \quad (7)$$

(c) The Prandtl number is assumed constant at 0.7.

(4) The properties of the coolant air were obtained from reference (11) based on a temperature at the injection slot exit of 530°R.

(5) The thermal conductivity of the nozzle wall material is given by reference (12). For the range 672°R  $\leq T_w \leq$  1212°R, the thermal conductivity is essentially constant at

$$k_w = 0.672 \text{ Btu-in/sec-ft}^2\text{-°R} \quad (8)$$

### Nozzle Heat Transfer Without Film Cooling

One-dimensional (radial) heat conduction through the nozzle wall was assumed. (This assumption is discussed more fully in a later section.) Thus

$$q = k_w (T_{wj} - T_{wo}) / r_i \log(r_o/r_j) \quad (9)$$

The heat transfer coefficient is, by definition

$$h_{\infty} = q / (T_{aw} - T_{wi}) \quad (10)$$

The adiabatic wall (recovery) temperature,  $T_{aw}$ , in equation (10) was evaluated based on a turbulent recovery factor of 0.89 and a specific heat ratio of 1.4:

$$T_{aw} = T_S \frac{1 + 0.178 M_{\infty}^2}{1 + 0.2 M_{\infty}^2} \quad (11)$$

From the nozzle geometry and measured values of  $T_{wj}$  and  $T_{wo}$ , the nozzle heat transfer results were obtained from equations (9) and (10), and are shown in figure 6 as  $(Nu_d)_{\infty}$  vs.  $(Re_d)_{\infty}$ , where

$$(Nu_d)_{\infty} = h_{\infty} d / k_{\infty} \quad (12)$$

$$(Re_d)_{\infty} = 4 \dot{m}_g^0 / \pi d \mu_{\infty} \quad (13)$$

Also included in figure 6 are the relation for turbulent pipe flow of Dittus and Boelter (ref. (13))

$$(Nu_d)_{\infty} = 0.023 (Re_d)_{\infty}^{0.8} (Pr)_{\infty}^{0.4} \quad (14)$$

and the relation for solid propellant rocket nozzle heat transfer of Colucci (ref. (14))

$$(Nu_d)_{\infty} = 0.023 (Re_d)_{\infty}^{0.8} \quad (15)$$

For laminar pipe flow, reference (15) gives

$$(Nu_d)_{\infty} = 4.36 \text{ for uniform heat flux} \quad (16)$$

$$(Nu_d)_{\infty} = 3.66 \text{ for uniform wall temperature}$$



Film-Cooled Nozzle Wall Temperatures

The equation of reference (7) giving the film-cooling effectiveness,  $\eta$ , for an adiabatic plate, is

$$\eta = \cos(0.88)e^{-\varphi} \quad (17)$$

where

$$\eta \equiv (T_{aw} - T_{wi}) / (T_{aw} - T_c) \quad (18)$$

$$\beta \equiv \tan^{-1} \left[ \frac{\sin \epsilon}{\cos \epsilon + (\rho u)_g / (\rho u)_c} \right] \quad (19)$$

$$\varphi \equiv \left[ \frac{h_\infty^0 L z}{(\dot{m} c_p)_c} - 0.04 \right] \left[ \frac{S u_g}{d_c} \right]^{1/8} f \quad (20)$$

The velocity ratio function,  $f$ , in equation (20) is defined by

$$f = 1 + 0.4 \tan^{-1} [(u_g/u_c) - 1] \quad (21)$$

when

$$(u_g/u_c) \leq 1.0$$

and

$$f = \left[ \frac{u_c}{u_g} \right]^{1.5} [(u_c/u_g) - 1] \quad (22)$$

when  $(u_c/u_g) \geq 1.0$

Equations (21) and (22) are shown in figure 7.

In analogy with equation (18), the film cooling effectiveness for a nonadiabatic surface is defined by\*

$$\eta' \equiv (T_{wi}^0 - T_{wi}) / (T_{wi}^0 - T_c) \quad (23)$$

The film cooling data were plotted in figures 8a through 8h as  $\eta'$  vs.  $\varphi$ . The measured values of  $T_{wj}$  and  $T_{wi}^0$  were assumed equal to  $T_{wi}$  and  $T_{wi}^0$ , respectively, in the evaluation of the measured effectiveness,  $\eta'$ . The straight line in these figures is given by the equation

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\*For an insulated (adiabatic) surface,  $T_{wi}^0 = T_{aw}$ .

$$\eta' = e^{-\eta} \quad (24)$$

In the experiment, the slot angle factor,  $\cos(0.88)$ , was approximately unity for all the condition encountered. In both equation (24) and data, the parameter  $\phi$  was obtained from equation (20) with  $h_{w0}$  evaluated from equation (14) and at  $z=0$ , and the terms  $u_g$  and  $f$  evaluated at  $z=0$  (ref. (8)).

## DISCUSSION AND CONCLUSIONS

The one-dimensional heat transfer analysis, used for all the data without film cooling, is expected to be fairly accurate because of the slenderness of the nozzle configuration (see fig. 2a). A comparison between the one-dimensional analysis and an exact solution of a sample temperature field yielded negligible differences in the resulting heat transfer rates. The experimental Nusselt numbers were, therefore, computed by the one-dimensional method. The results showed fair agreement with the turbulent flow correlations as shown in figure 6. The scatter in the data of figure 6 can be attributed mainly to the relatively low wall temperature levels and temperature gradients. The probable error in the measured heat transfer rates (thus, the Nusselt numbers) was estimated to be as low as  $\pm 2\%$  in the throat region and as high as  $\pm 30\%$  at  $z=1.25''$  and  $z=9.0''$ .

The film-cooling effectiveness as defined by equation (23) is a measure of the decrease in the wall temperature due to film cooling relative to the maximum possible decrease, whereas the original definition (Eq. (18)) has no physical meaning when applied to a nonadiabatic surface. The heat transfer coefficient,  $h_{w0}$ , appearing in the parameter  $\phi$  of equation (20) was computed from equation (14). The correlation thus obtained (Eq. (24)) is shown in figures 8a through 8h. The scatter in the data is mainly due to the low measured wall temperatures. The ratio of the two small differences that defined the film-cooling effectiveness (Eq. (23)) can be quite inaccurate. Conversely, the relatively large discrepancies between data and equation (24) indicated in figures 8a through 8h resulted in only moderate discrepancies in terms of the wall temperatures. An example of this is shown by comparing the solid curves with the data in figure 9.\* The dashed curves in figure 9 will be discussed later.

The effect of film cooling on the nozzle throat temperature is shown in figure 10 along with the predictions of equation (24). The dashed curves in figure 10 are the estimated throat temperatures that would result if the coolant and the main stream were completely mixed at this point. Figures 8 to 10 will be discussed more fully later.

\*Only twelve data points appear in figure 9 instead of the designed fifteen because three thermocouples became inoperative during the experiment.

In an attempt to study the phenomenon of film cooling and, perhaps, to improve the correlation, the theoretical flow model of reference (6) was extended to the case of a nonadiabatic, axisymmetric nozzle (fig. 11). The assumptions used in the development of the extension are as follows:

- (1) The coolant does not mix with the main stream.
- (2) Heat conduction through the nozzle wall is one-dimensional radial.
- (3) The local coolant temperature is equal to the local wall temperature.
- (4) The temperature gradient through the coolant film is negligible. (These first four assumptions are the same as those of reference (6).)
- (5) Heat transfer from the main stream is governed by equations (10) and (11) with  $h_o^o$  computed from equation (14).
- (6) The water temperature is constant at 530°R. (The total rise in the water temperature from inlet to exit due to heat transfer from the nozzle wall was estimated to be less than 10°R for the worst case.)
- (7) The coolant temperature at injection slot exit is 530°R.

A heat balance  $Q_1 = Q_2 + Q_3$  and  $Q_3 = Q_4$  (fig. 11) yielded the following differential equation:

$$\frac{dT_{wi}}{dz} = \frac{2\pi}{(\dot{m} c_p)_c} \left[ h_o^o r_i (T_{aw} - T_{wi}) - \frac{T_{wi} - T_\infty}{(\log(r_o/r_i)/k_w) + (1/h_w r_o)} \right] \quad (25)$$

in which  $h_o^o$ ,  $r_i$ , and  $T_{aw}$  were all allowed to vary with  $z$ . Typical numerical solutions of equation (25) are shown in figure 9 as dashed curves. It is seen from figure 9 that the extension did not offer a better correlation of film-cooling data than equation (24). Thus, it is concluded that the phenomenon of film cooling of a nonadiabatic nozzle is more complex than that described by equation (25).

Although no realistic flow model was developed for the film cooling of a nozzle, the following qualitative observations may be made:

(1) Figures 8a through 8h indicate that equation (24) generally overestimates the effectiveness for the subsonic flow regime and underestimates it for the sonic and the supersonic flow regimes. Thus, equation (24) would be conservative in estimating the film-cooled nozzle throat temperatures (fig. 10).

(2) Although figures 8a through 8h showed large discrepancies between the measured and the predicted film-cooling effectivenesses, the difference between the measured and the predicted wall temperatures is less severe.

(3) The throat temperature (fig. 10) decreases with increasing coolant flow until an optimum value is reached. Further increase in the coolant flow, in some cases, resulted in an increase in the throat temperature. This reversal is believed to be due to the premature mixing of the coolant with the main stream, resulting in a loss of the insulation effect.

(4) Optimum cooling is achieved when the coolant velocity at the injection slot exit is approximately equal to the main stream velocity at that point. This can be seen from figure 10 and the following table:

$T_s$	$\left[ \frac{\dot{m}_c}{\dot{m}_g} \right]_{(u_c/u_g)_{z=0} = 1}$
672°R	4.5%
852°R	5.6%
1032°R	6.8%
1212°R	8.0%

For lack of a rigorous and exact analysis and a comprehensive experimental investigation, it is felt that, for engineering purposes, equation (24) offers a fair correlation of film-cooling data and may be used in estimating the film-cooled wall temperatures.

#### SUMMARY

The air-film cooling of a Mach 2.4, nonadiabatic wall, axially symmetric, contoured nozzle was investigated experimentally. The main stream was air at supply conditions of 672°R to 1212°R and 115 psia to 465 psia. The film coolant was air at ambient temperatures and injected through a single annular slot of 10°

discharge angle. The coolant to main stream mass flow ratios were varied up to 20 percent. Steady-state temperature distributions along the inner and the outer walls were measured in both the subsonic and the supersonic flow regimes.

For the supply conditions tested, the main stream flow was fully developed and turbulent. The heat transfer data without film cooling were correlated reasonably well with the equation of Dittus and Boelter for turbulent pipe flow.

The film-cooled nozzle wall temperatures were correlated qualitatively with a modified version of the equation of Hatch and Papell, the modification being a definition of the film-cooling effectiveness for a nonadiabatic surface.

Optimum cooling is achieved when the velocities of the two streams at the injection slot exit are approximately equal.

For engineering purposes, the film-cooled wall temperatures may be estimated by the following procedure (for both adiabatic and nonadiabatic surfaces):

(1) Calculate, by conventional methods, the wall temperatures in the absence of film cooling.

(2) For a given coolant flow rate, compute the effectiveness from equation (17) or (24), evaluating all the parameters (except "z") in the term  $\varphi$  at  $z=0$ .

(3) Compute the film-cooled wall temperature from equation (18) or (23).

Steps (2) and (3) are iterated if the film-cooled wall temperature is prescribed and the coolant flow rate is to be determined.

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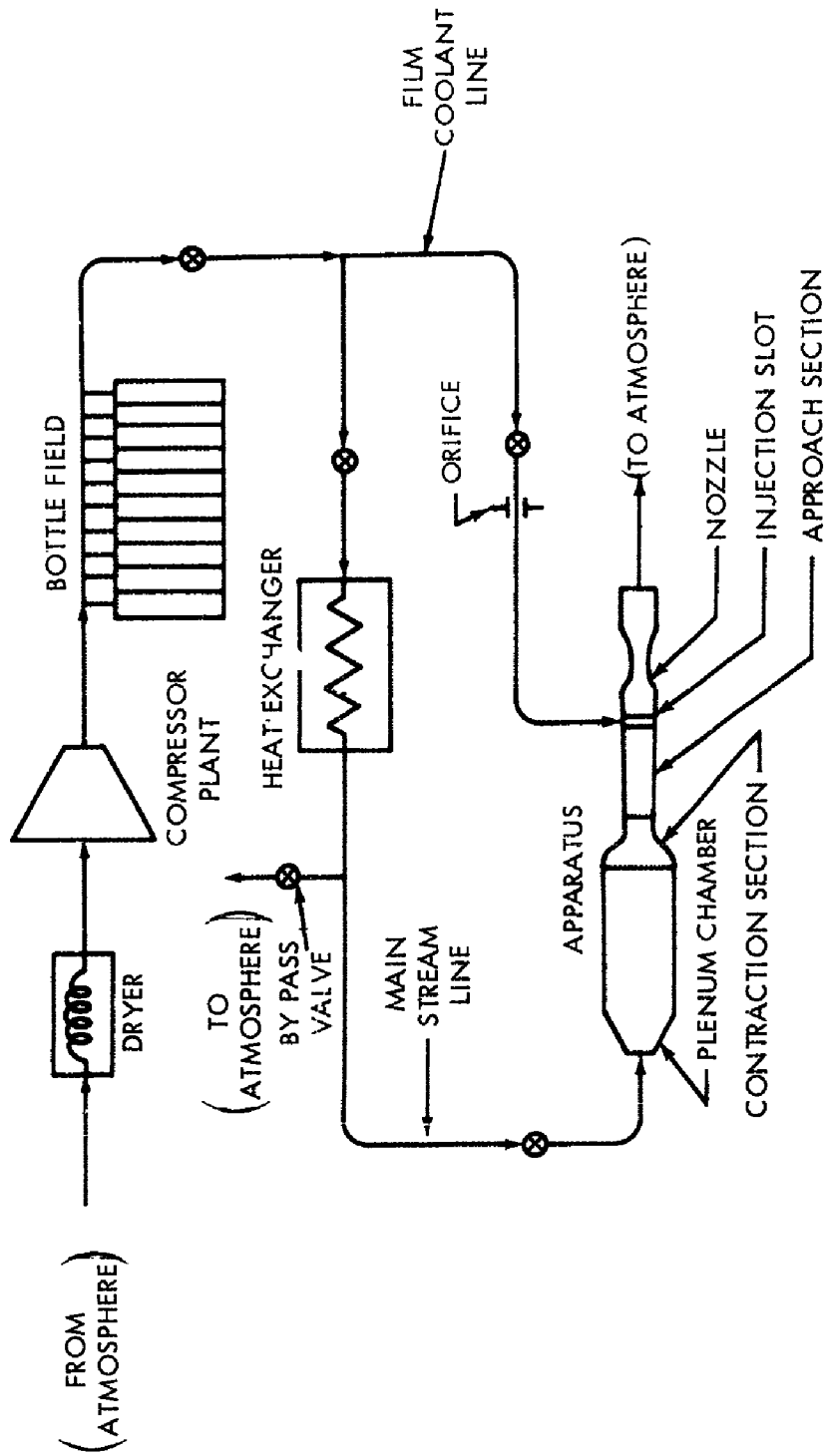


FIG. 1. FILM-COOLING APPARATUS SCHEMATIC



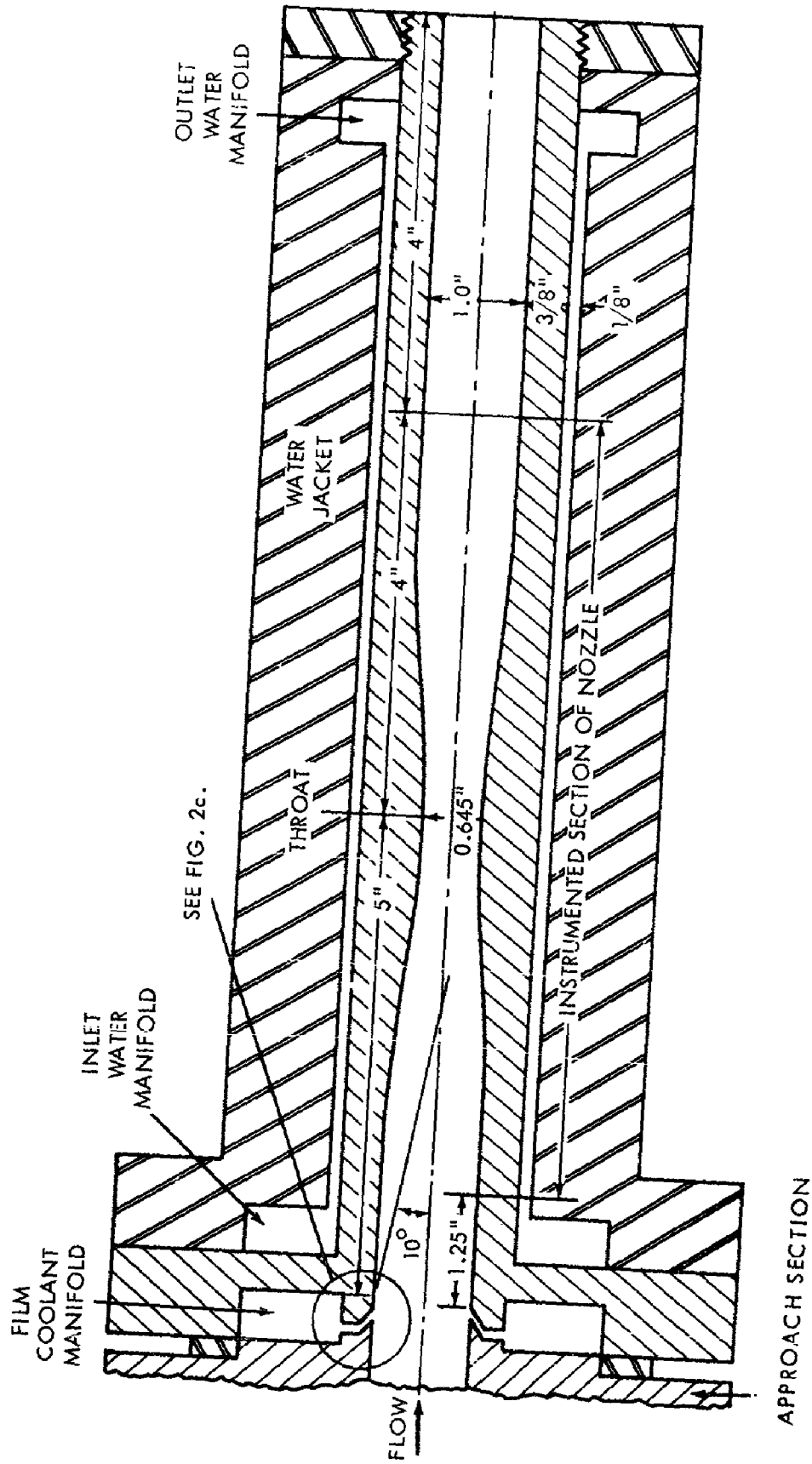


FIG. 2a. NOZZLE CONFIGURATION

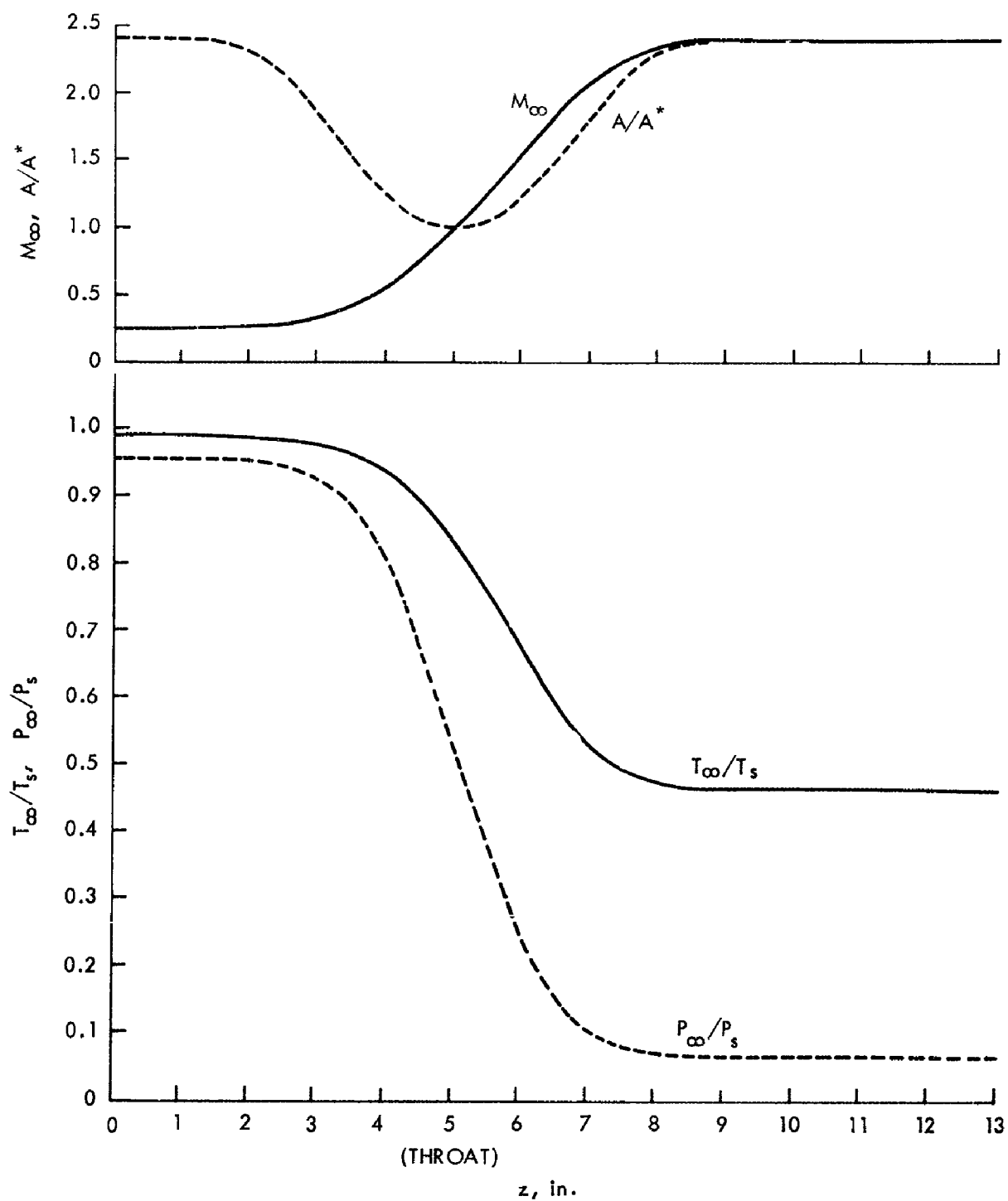


FIG. 2b. DESIGN FLOW CHARACTERISTICS OF NOZZLE

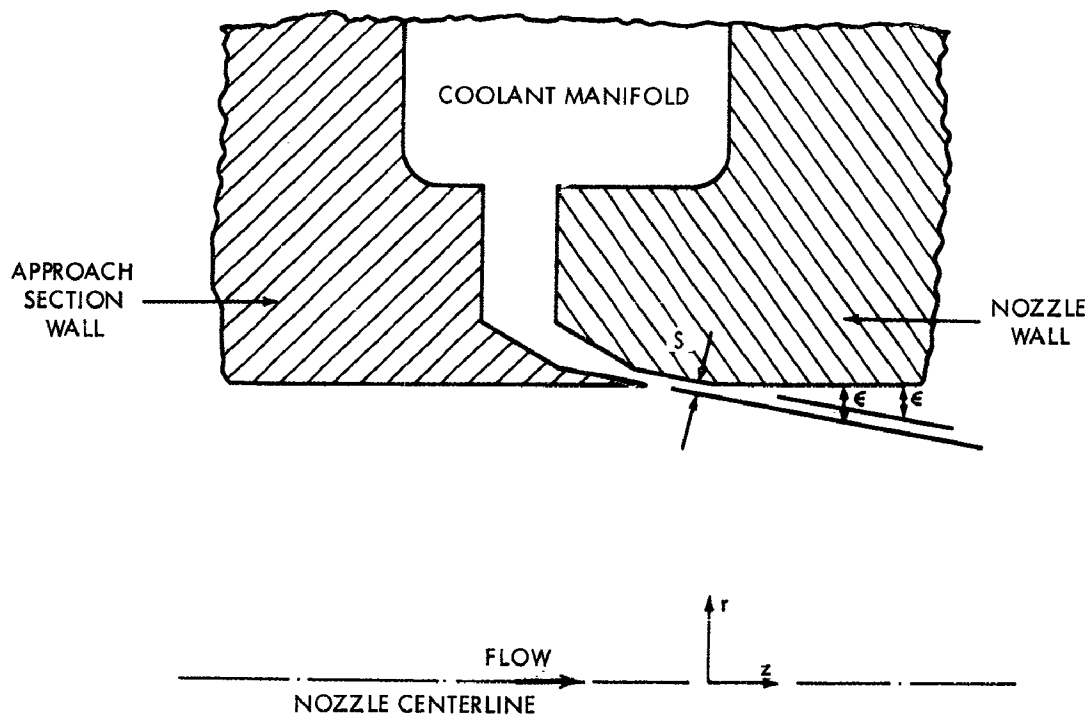


FIG. 2c. COOLANT INJECTION SLOT

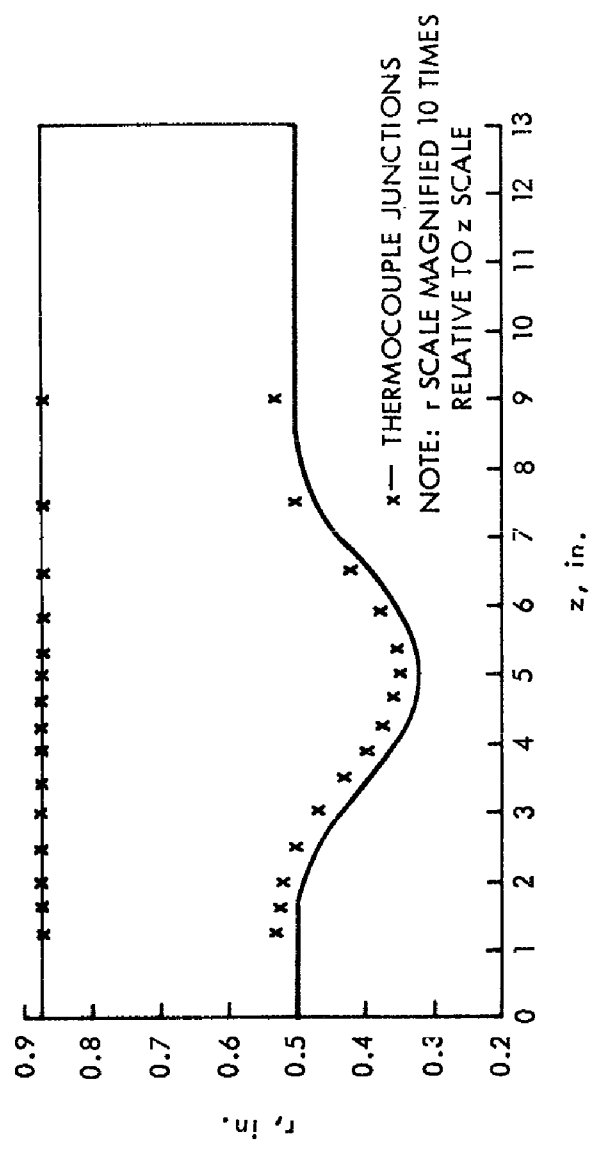


FIG. 3. THERMOCOUPLE LOCATIONS

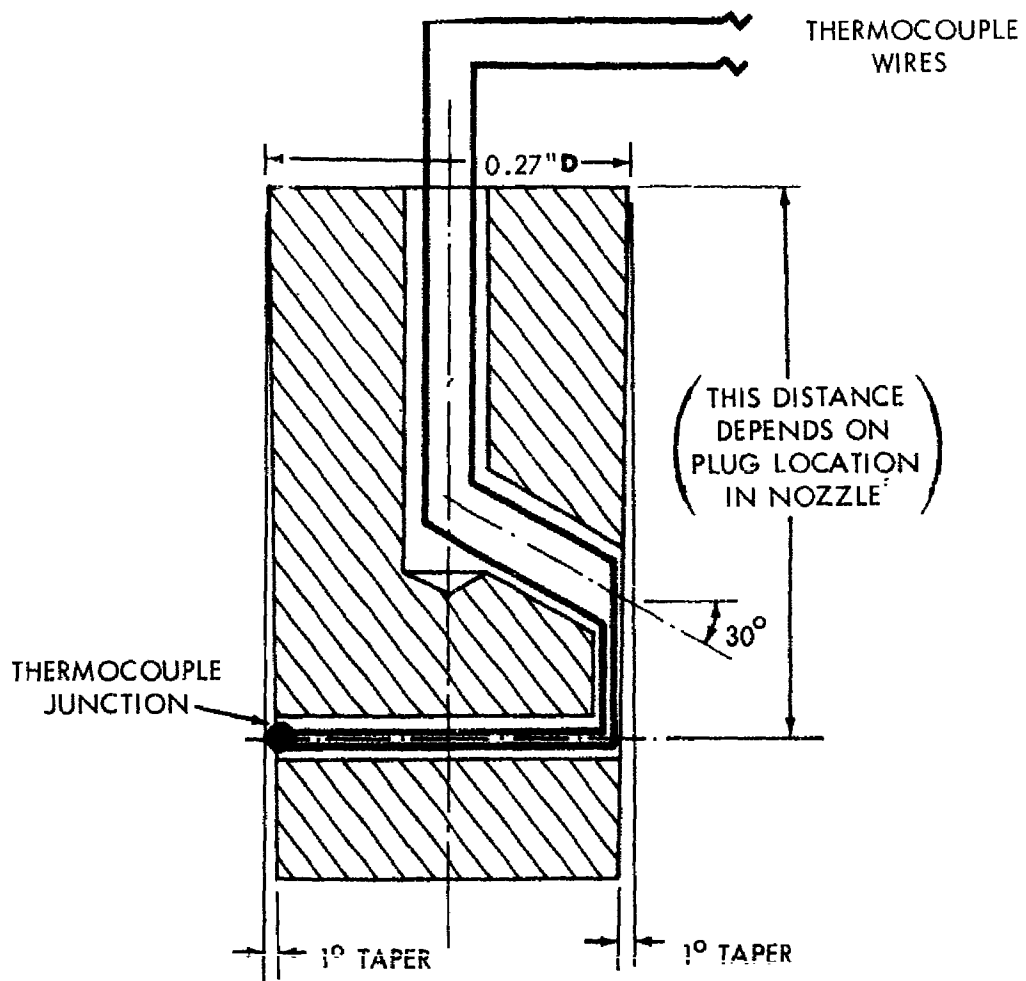


FIG. 4. THERMOCOUPLE PLUG DESIGN

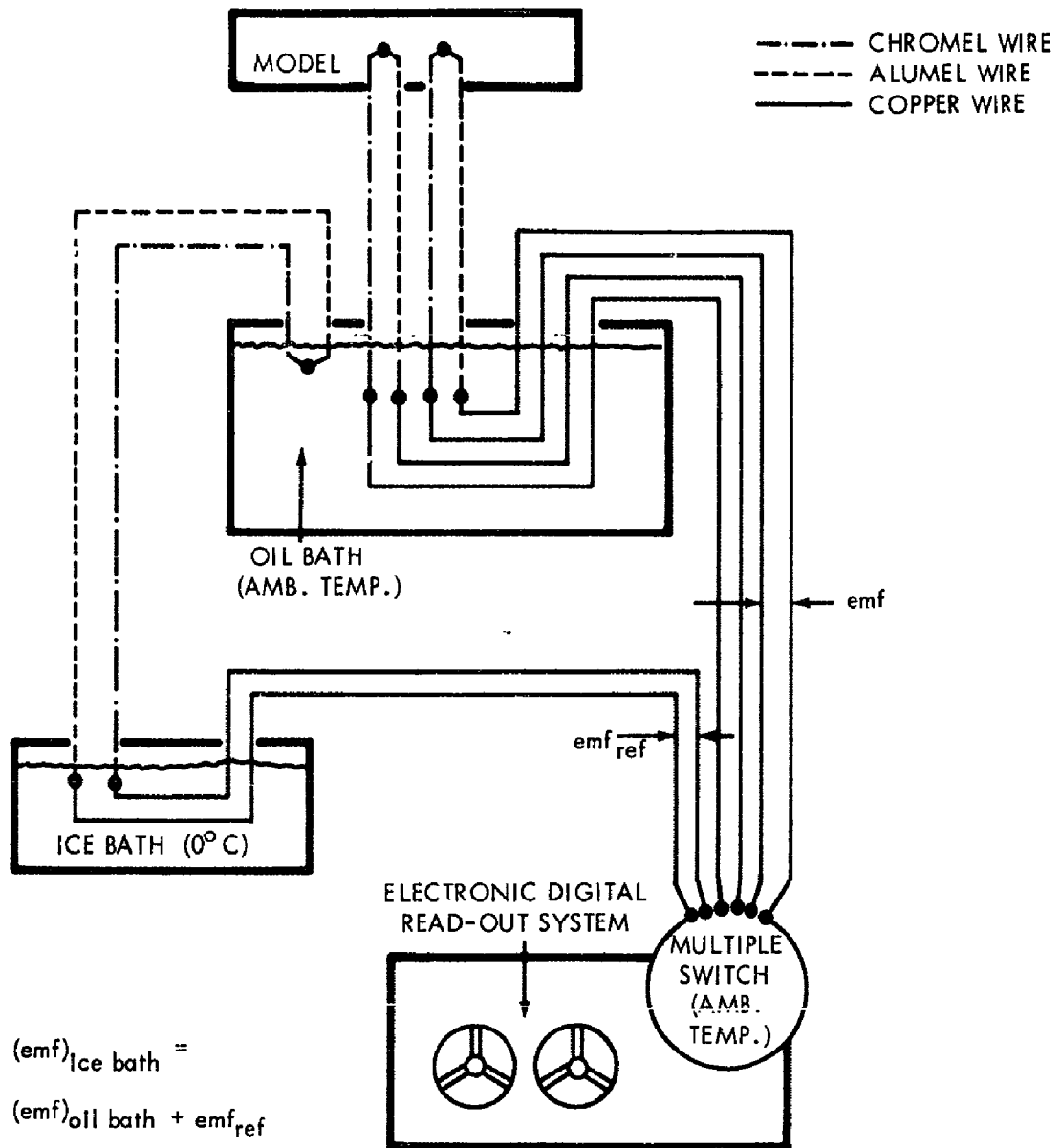


FIG. 5. THERMOCOUPLE READ-OUT SCHEMATIC

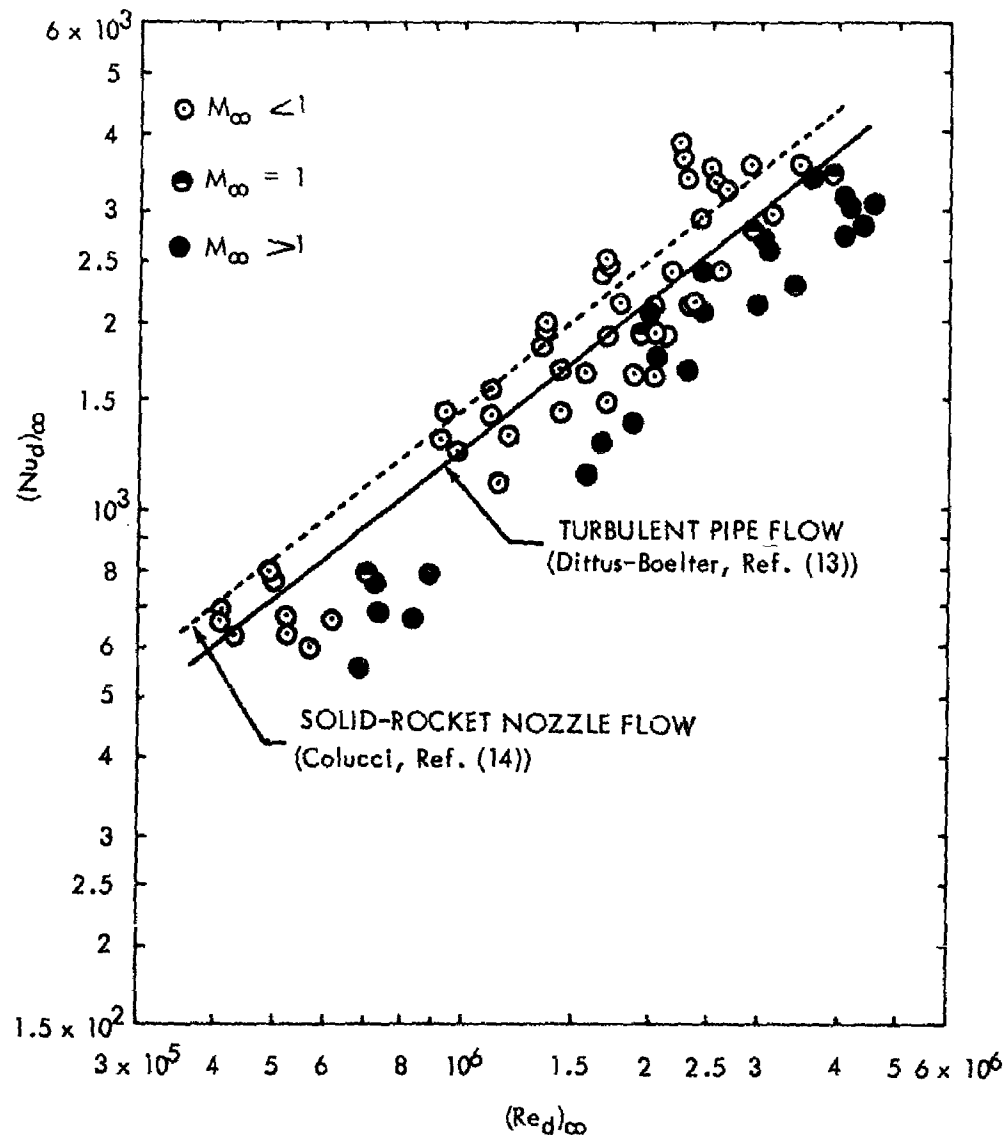


FIG. 6. NOZZLE HEAT-TRANSFER CORRELATION

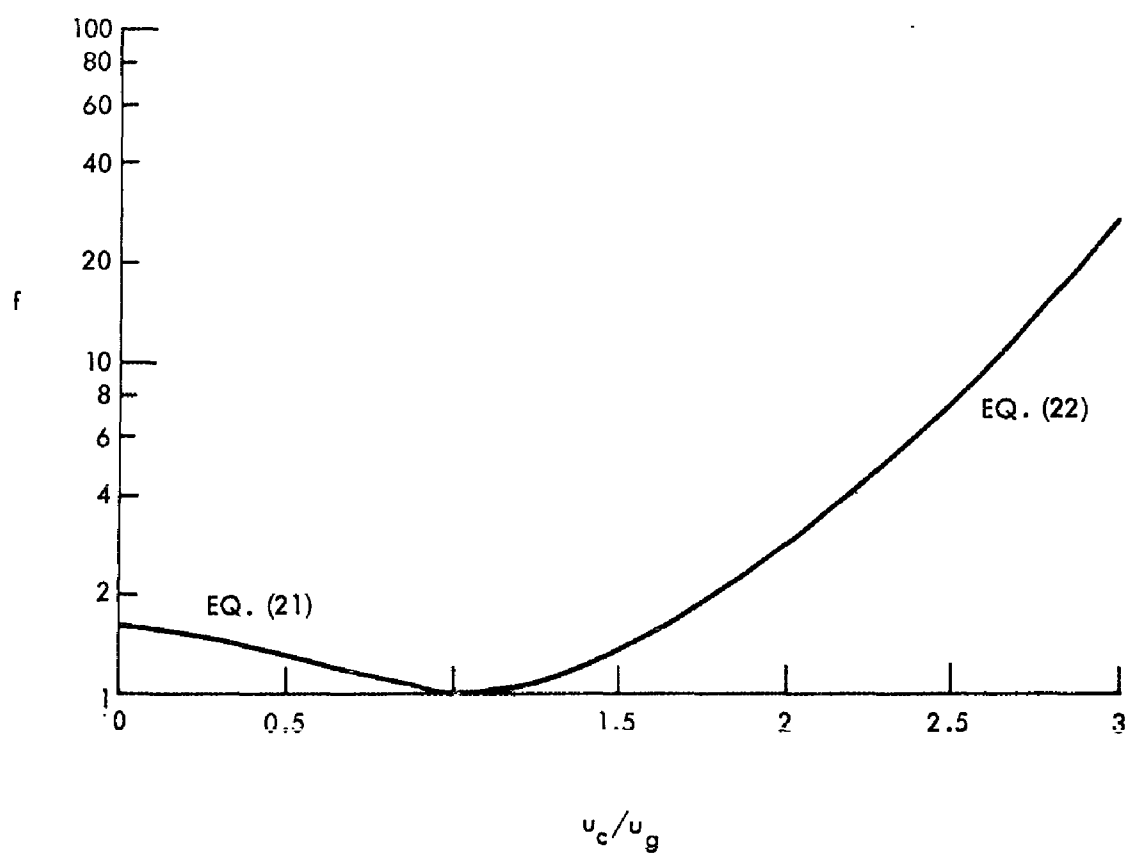


FIG. 7. VELOCITY RATIO FUNCTION



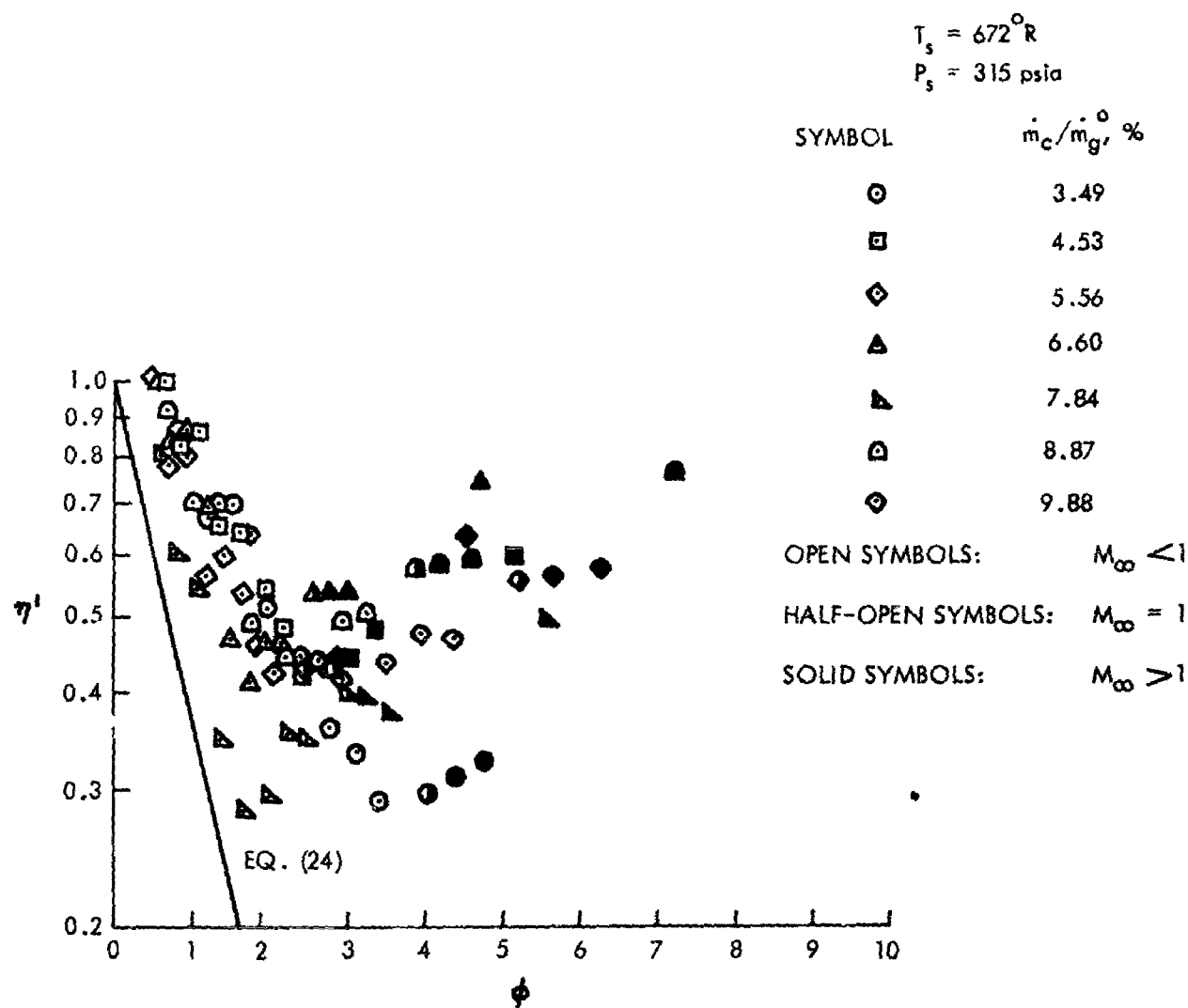


FIG. 8a. NOZZLE FILM-COOLING CORRELATION  
FOR  $T_s = 672^\circ\text{R}$  AND  $P_s = 315 \text{ psia}$

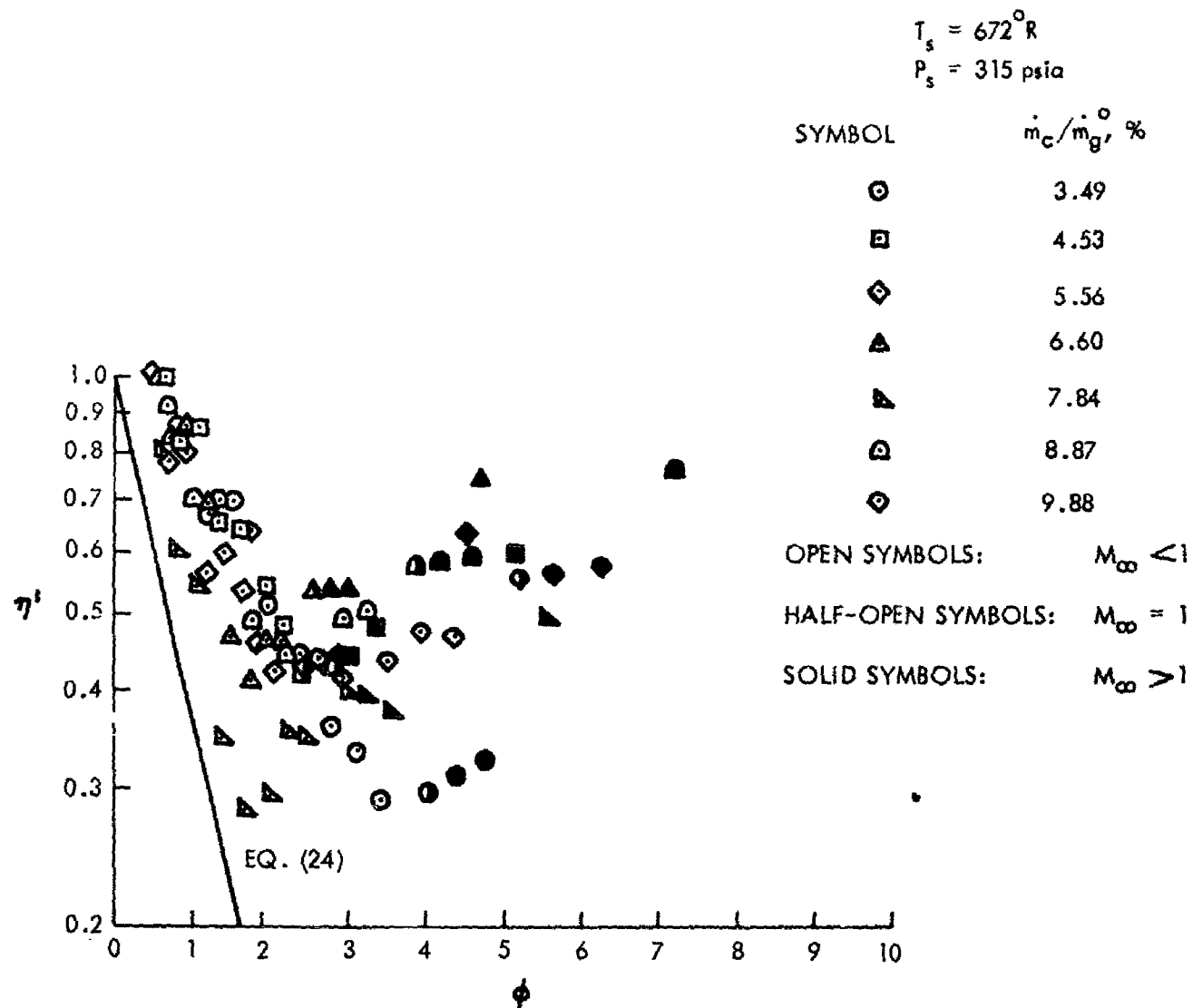


FIG. 8a. NOZZLE FILM-COOLING CORRELATION  
FOR  $T_s = 672^\circ\text{R}$  AND  $P_s = 315 \text{ psia}$

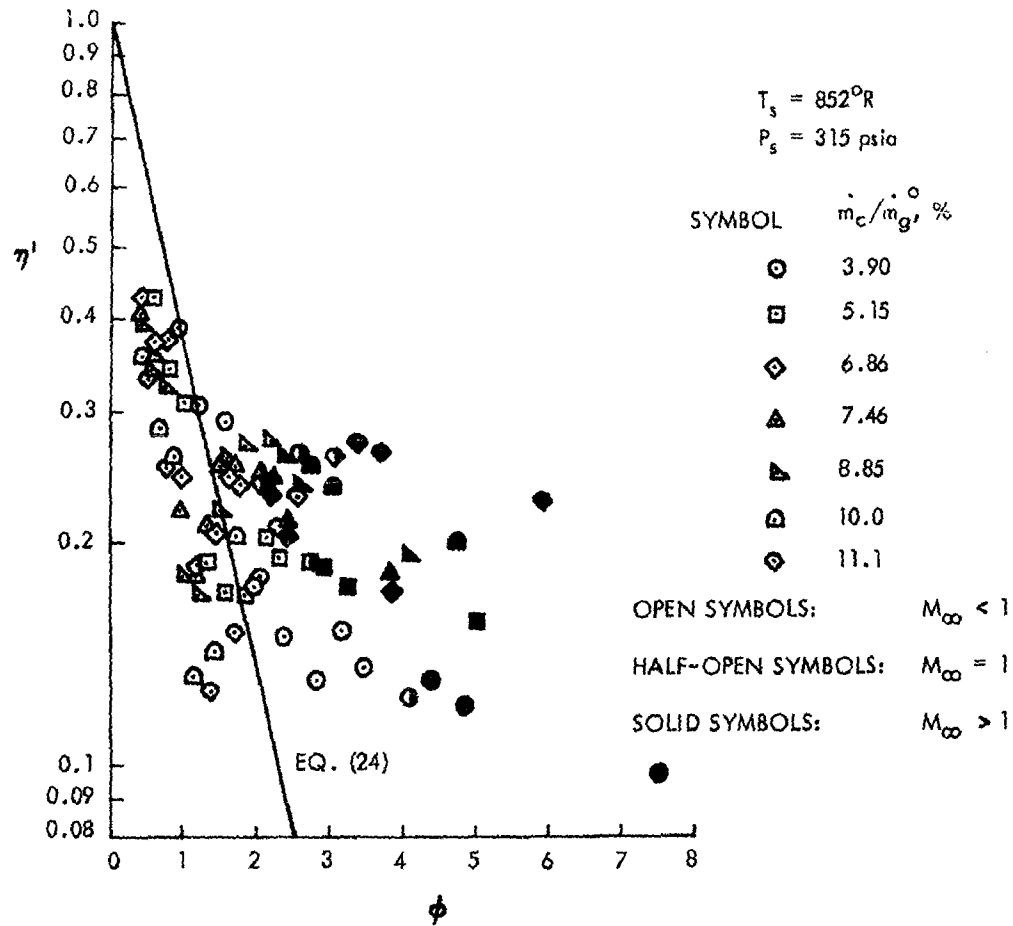


FIG. 8b. NOZZLE FILM-COOLING CORRELATION  
 FOR  $T_s = 852^\circ\text{R}$  AND  $P_s = 315 \text{ psia}$

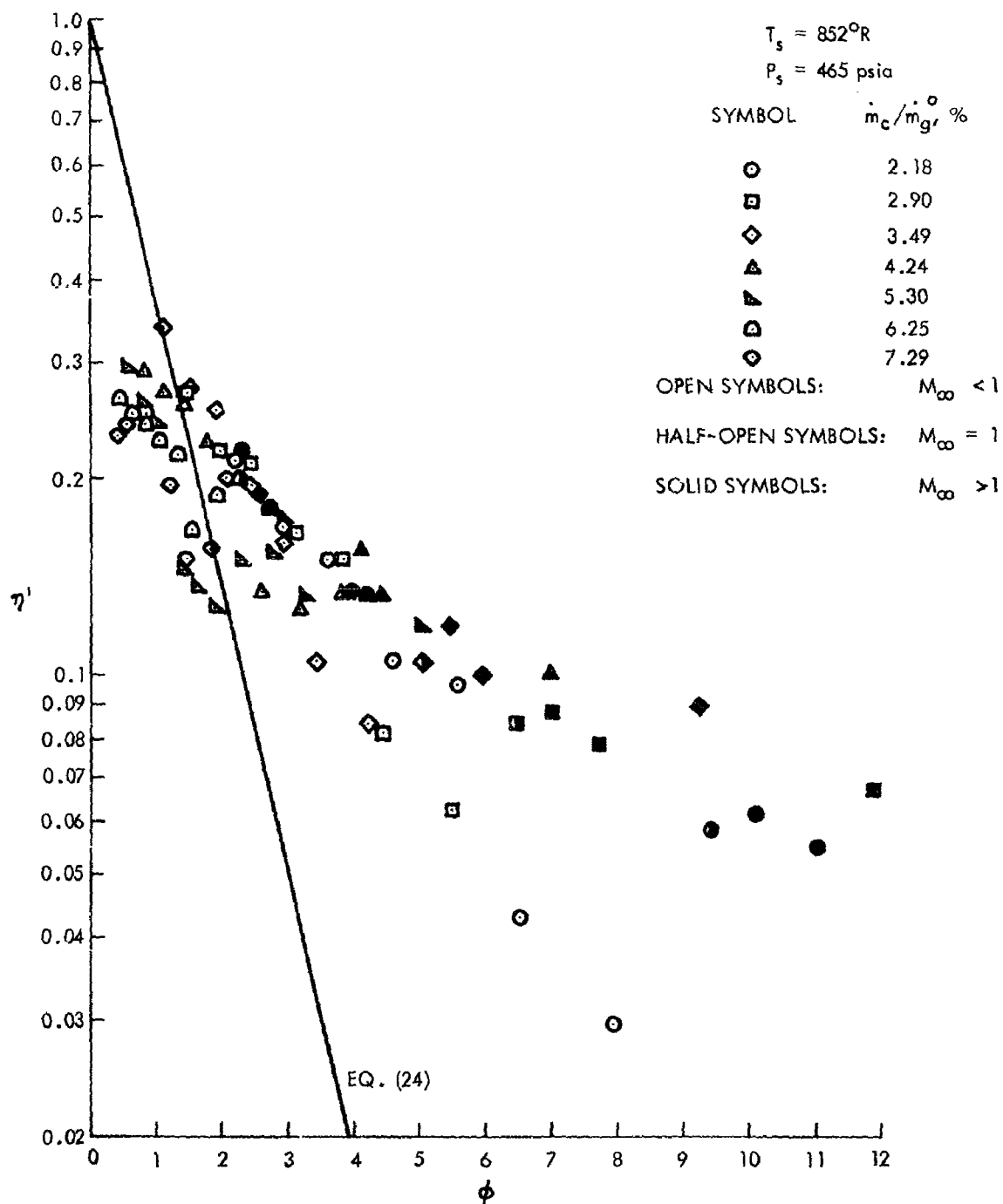


FIG. 8c. NOZZLE FILM-COOLING CORRELATION  
 FOR  $T_s = 852^\circ\text{R}$  AND  $P_s = 465 \text{ psia}$

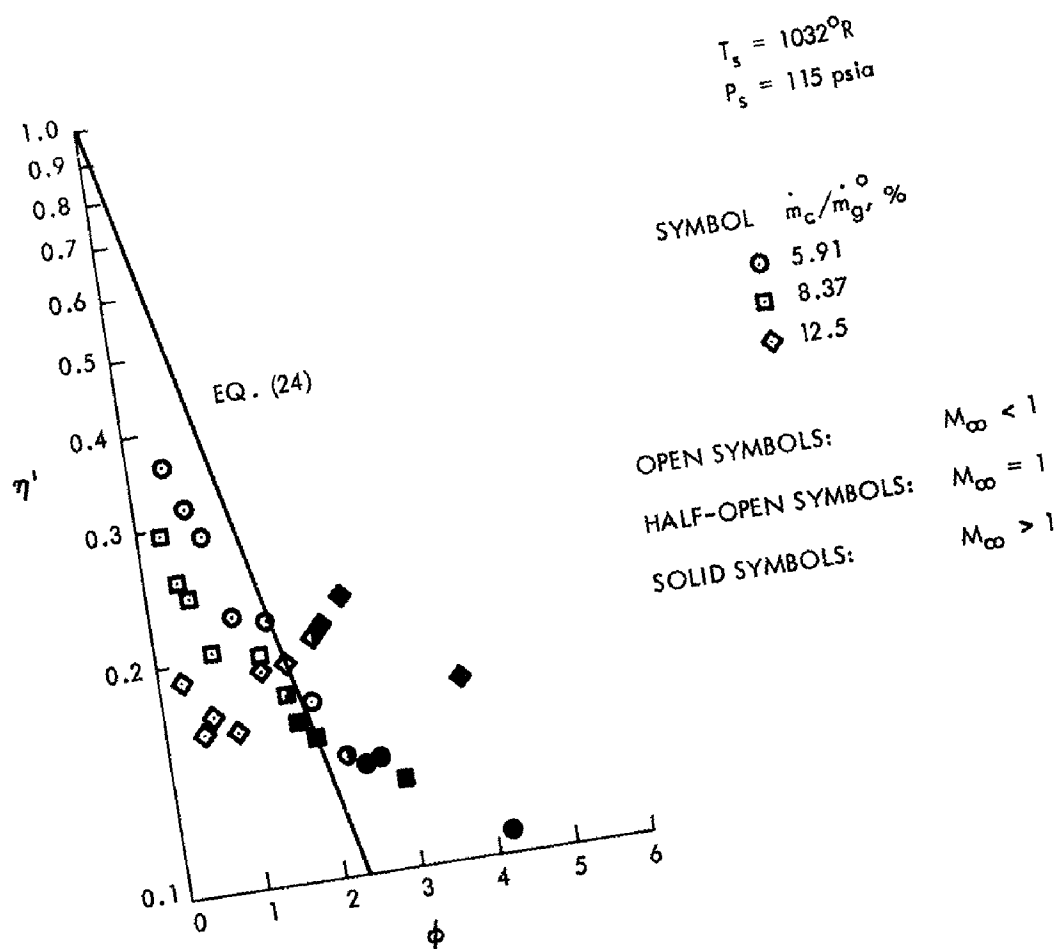


FIG. 8d. NOZZLE FILM-COOLING CORRELATION  
 FOR  $T_s = 1032^\circ\text{R}$  AND  $P_s = 115 \text{ psia}$

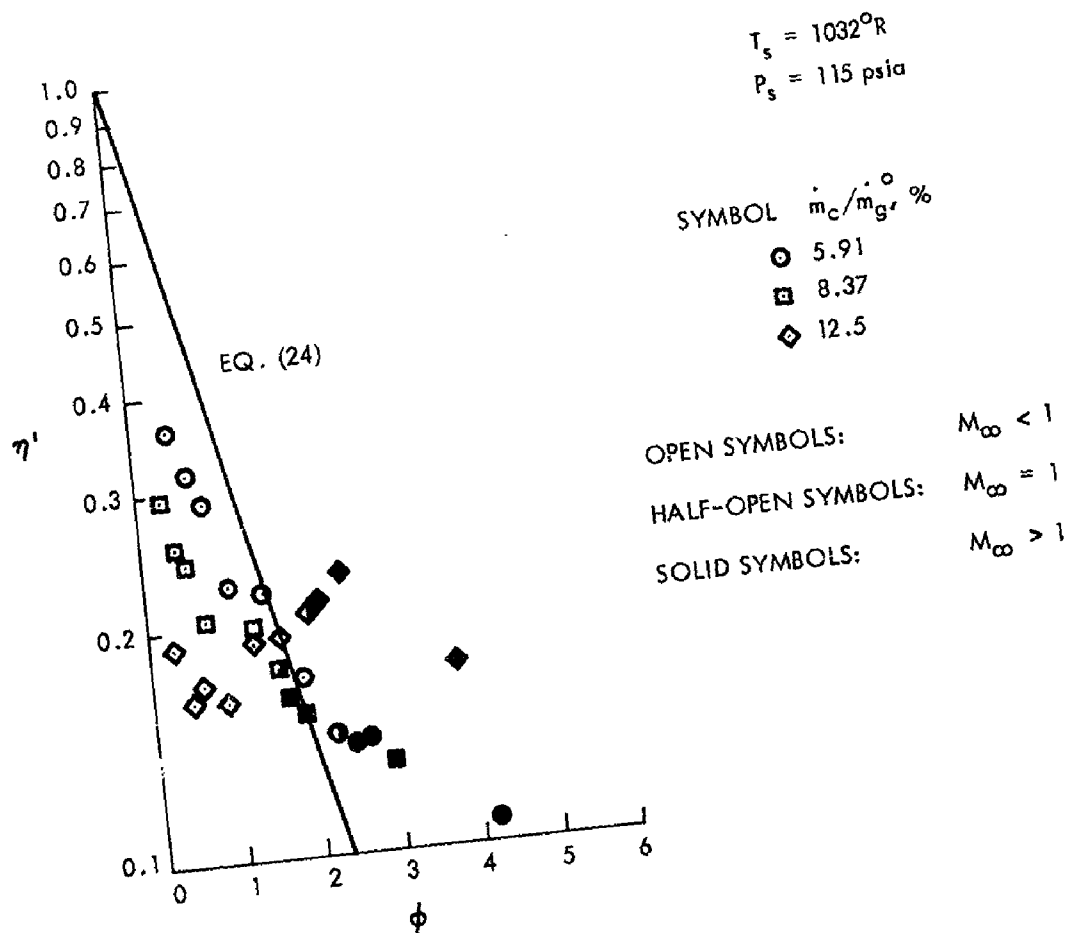


FIG. 8d. NOZZLE FILM-COOLING CORRELATION  
 FOR  $T_s = 1032^\circ\text{R}$  AND  $P_s = 115 \text{ psia}$

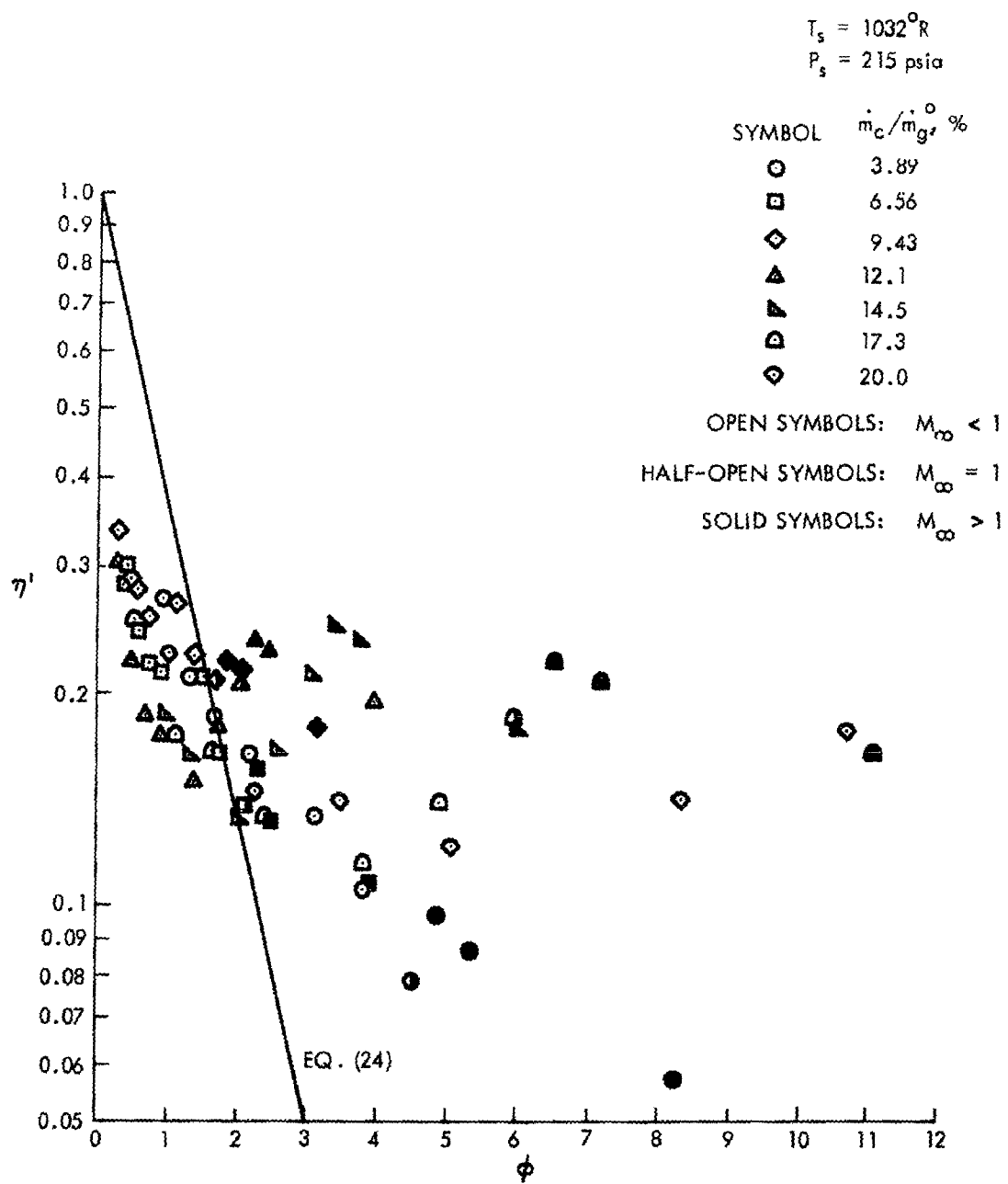


FIG. 8e. NOZZLE FILM-COOLING CORRELATION  
FOR  $T_s = 1032^\circ\text{R}$  AND  $P_s = 215 \text{ psia}$

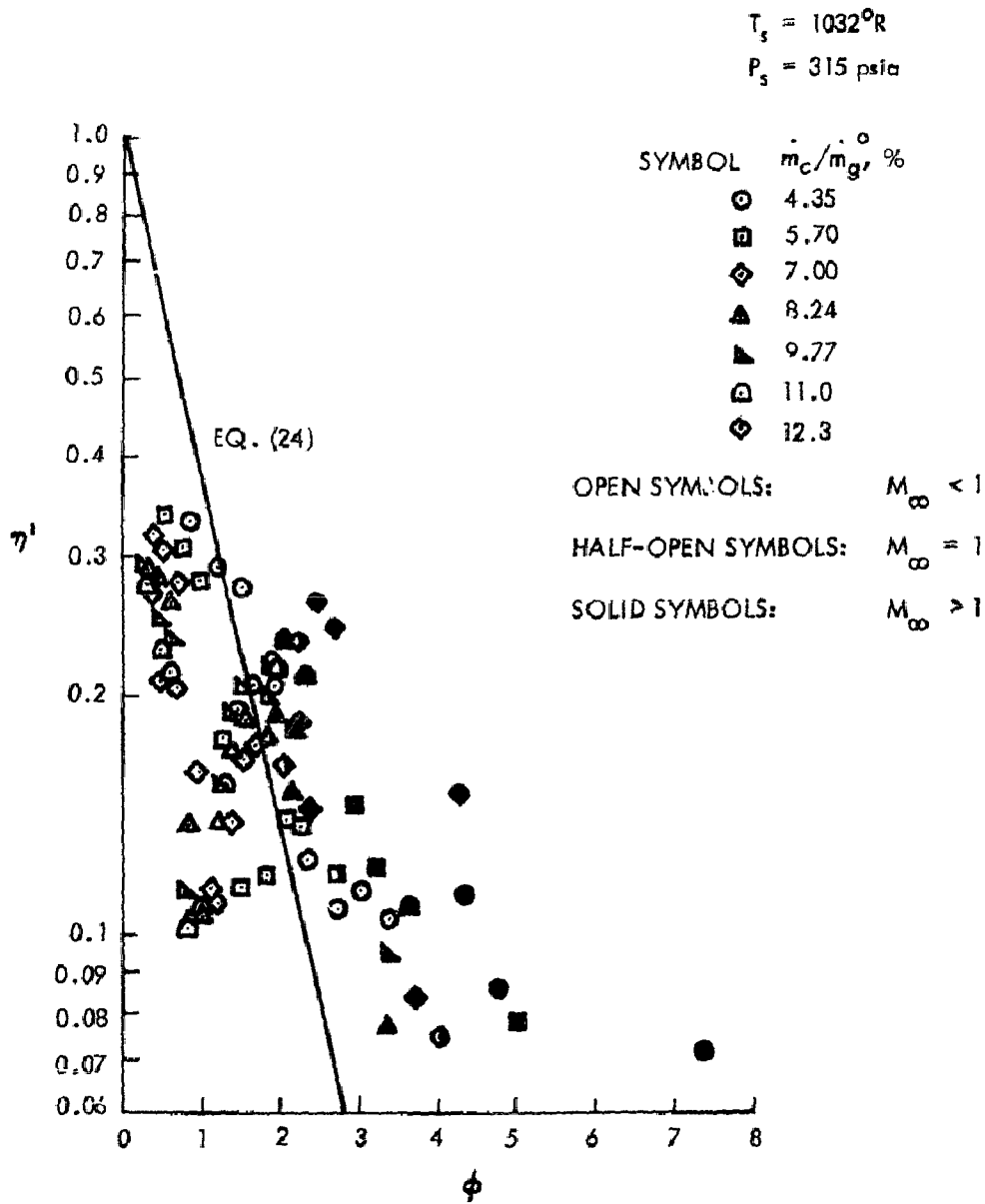


FIG. 8f. NOZZLE FILM-COOLING CORRELATION  
FOR  $T_s = 1032^\circ\text{R}$  AND  $P_s = 315 \text{ psia}$



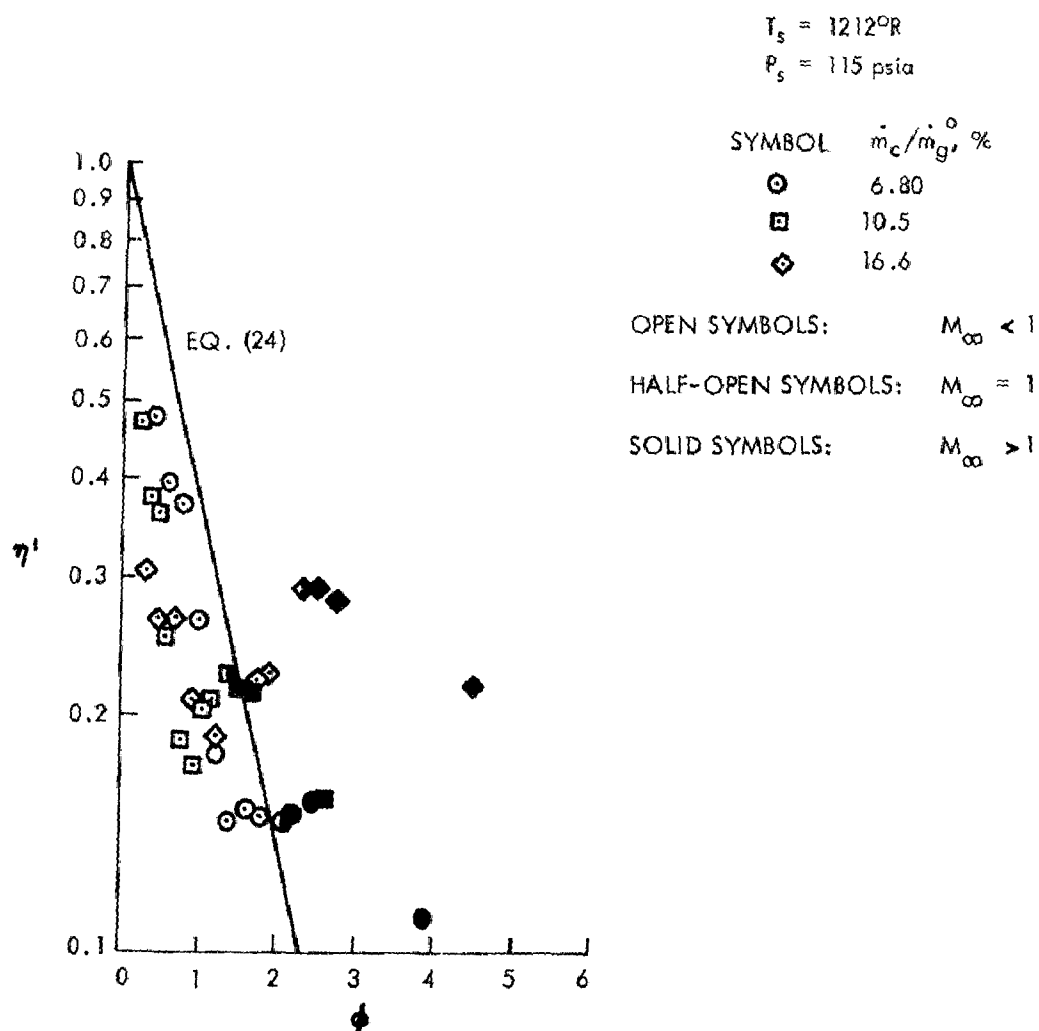


FIG. 8g. NOZZLE FILM-COOLING CORRELATION

FOR  $T_s = 1212^\circ\text{R}$  AND  $P_s = 115 \text{ psia}$

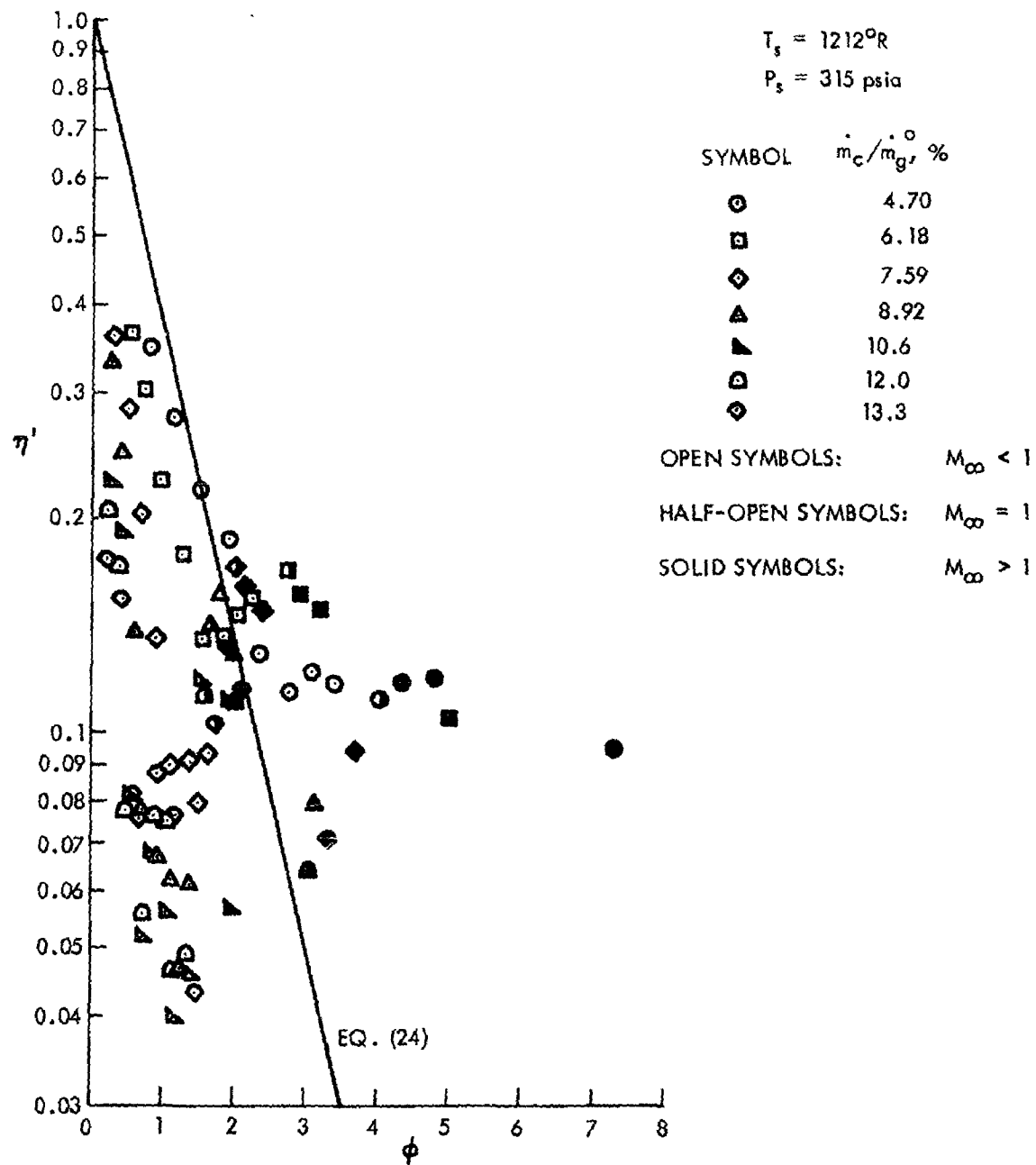


FIG. 8h. NOZZLE FILM-COOLING CORRELATION  
FOR  $T_s = 1212^\circ\text{R}$  AND  $P_s = 315 \text{ psia}$

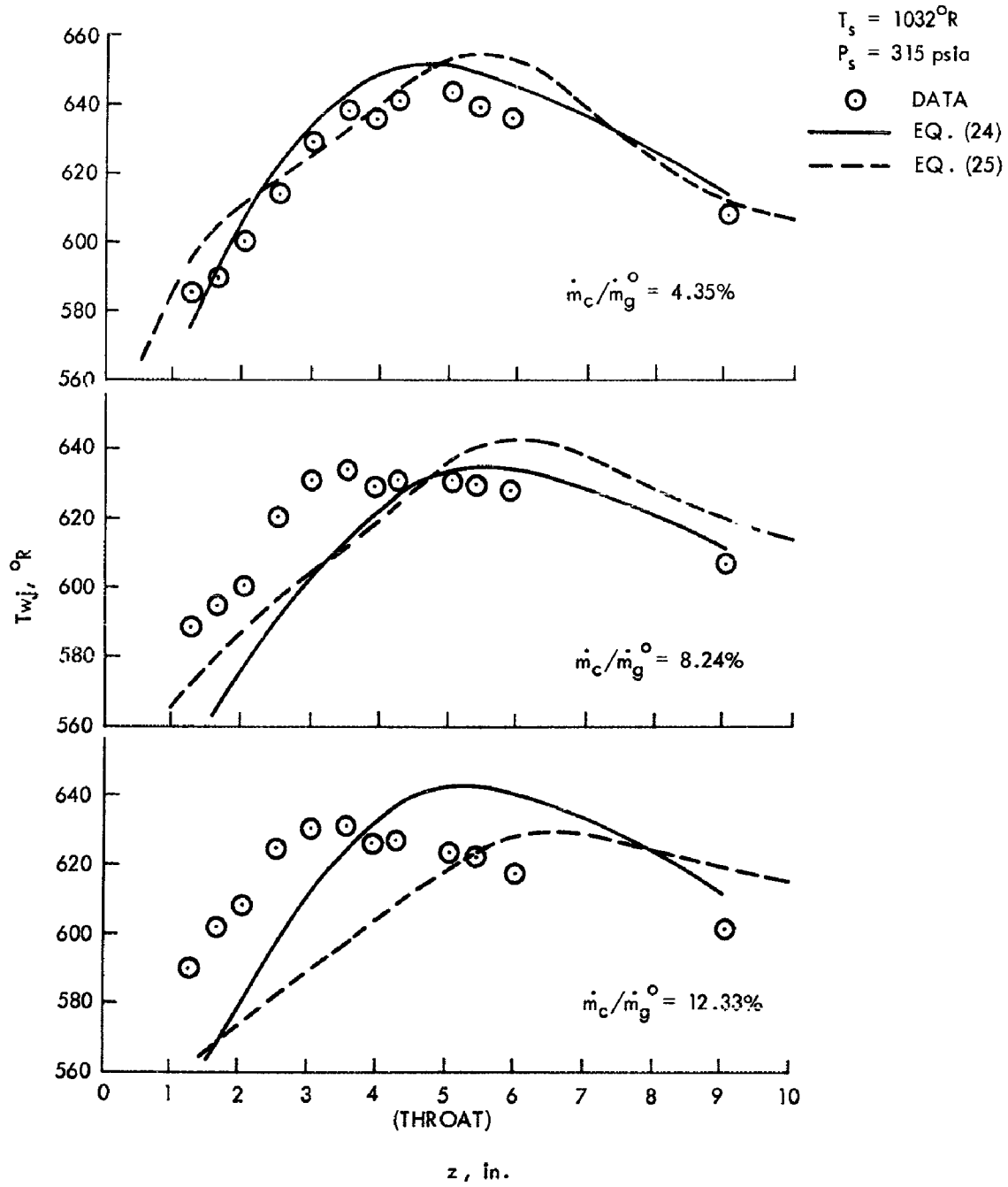


FIG. 9. MEASURED AND PREDICTED WALL TEMPERATURES

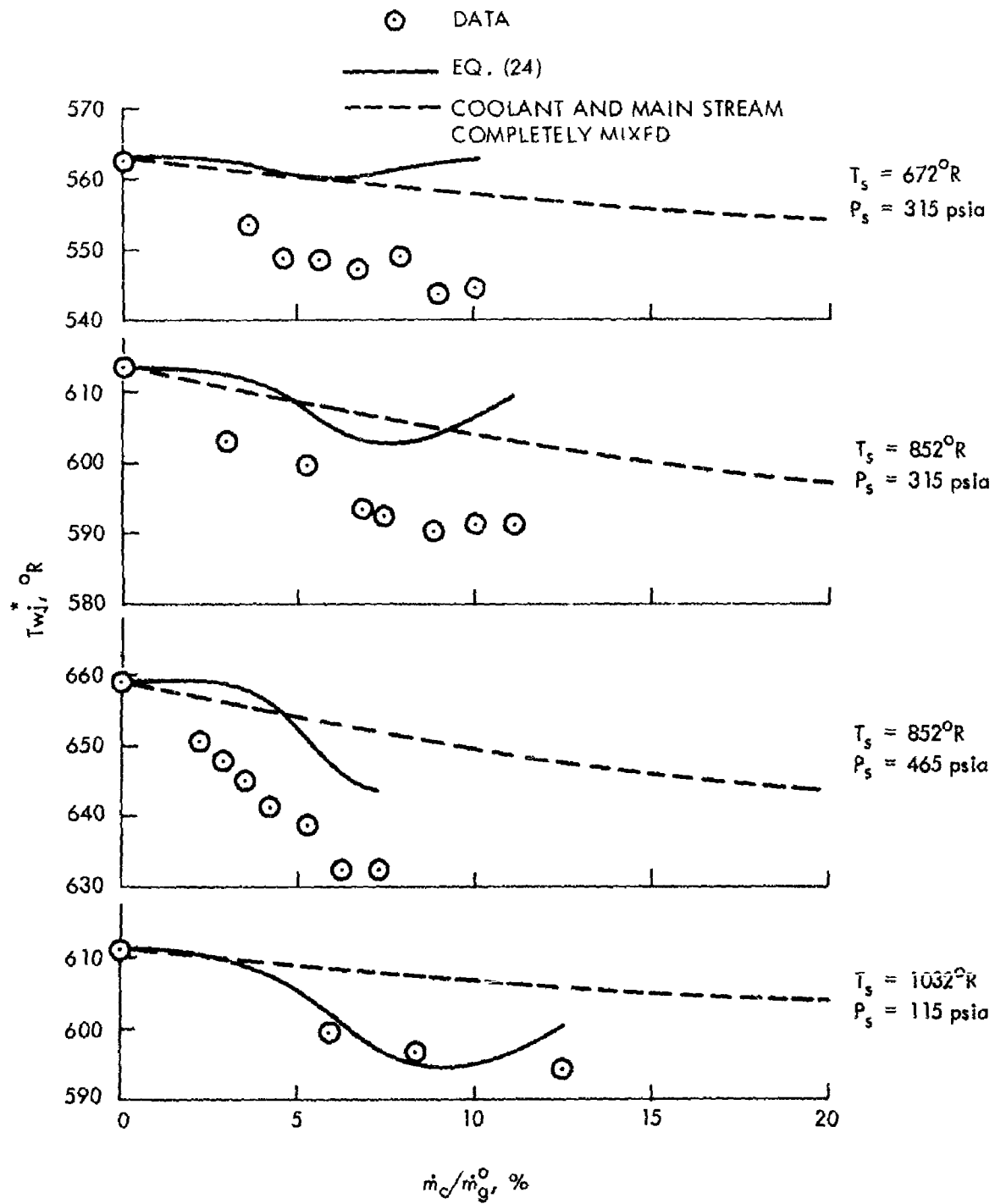


FIG. 10. FILM-COOLED NOZZLE THROAT TEMPERATURES

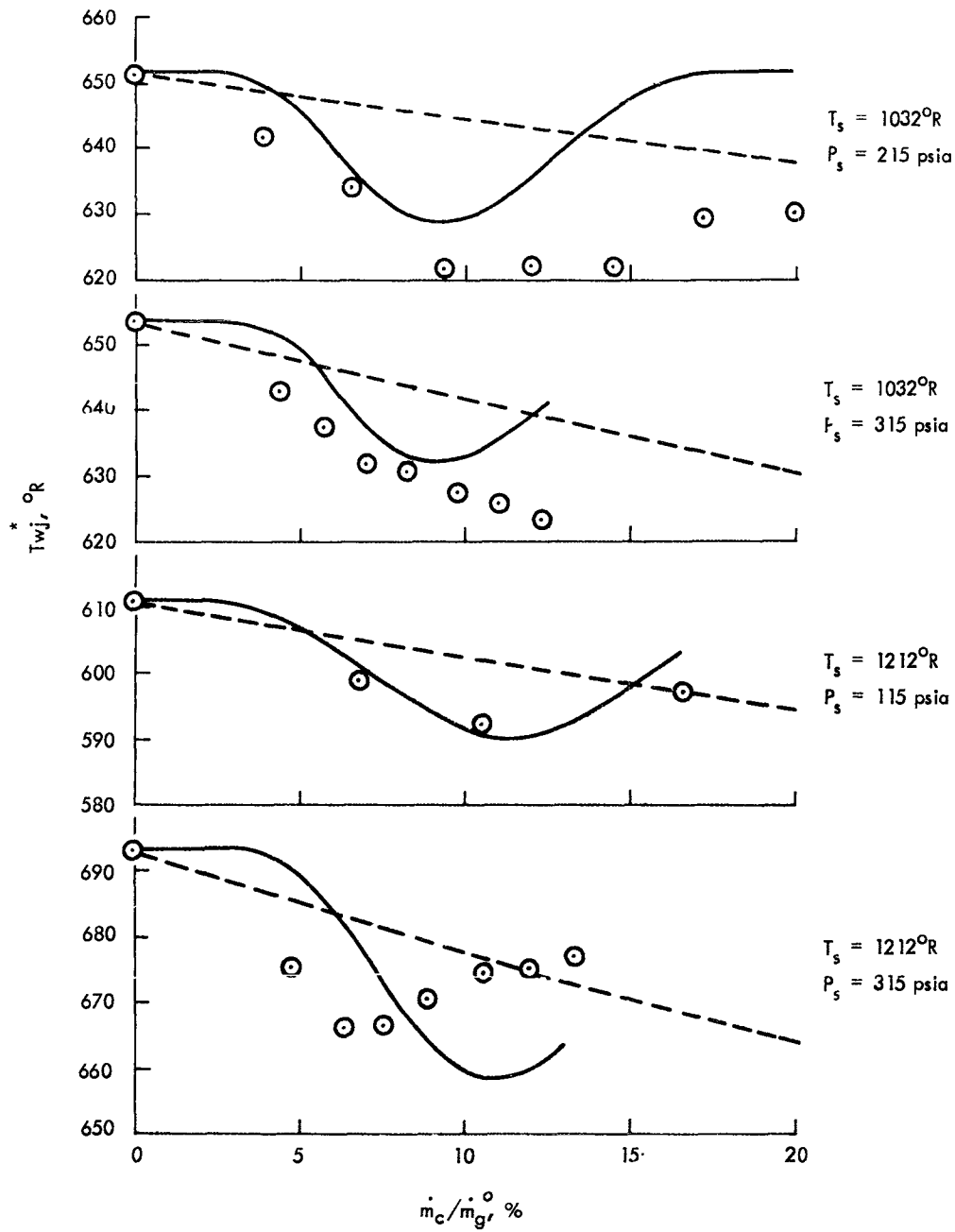


FIG. 10. (CONT'D) FILM-COOLED NOZZLE THROAT TEMPERATURES

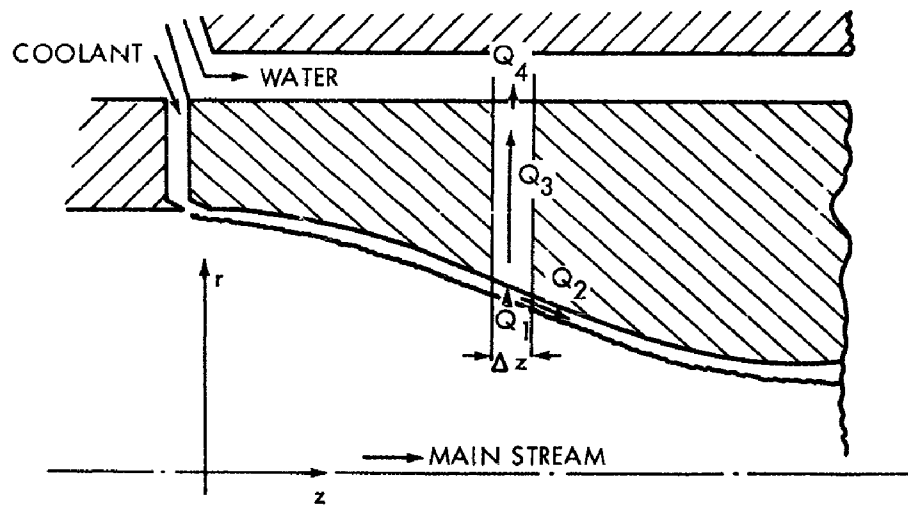


FIG. 11. EXTENDED FLOW MODEL OF REFERENCE (6)

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	DESCRIPTORS	CODES	SECURITY CLASSIFICATION AND CODE COUNT	DESCRIPTORS	CODES
SOURCE	NOL technical report	NOLTR		Unclassified - 36	U036
REPORT NUMBER	64-65	640065	CIRCULATION LIMITATION		
REPORT DATE	18 August 1964	0864	CIRCULATION LIMITATION OR BIBLIOGRAPHIC		
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## SUBJECT ANALYSIS OF REPORT

	DESCRIPTORS	CODES	DESCRIPTORS	CODES
Air		AIRE	High temperature	HTEM
Film		COAT	Temperature	TEMP
Cooling		COOL	Supply	SUPP
Nozzle		NOZZ	Pressures	PRES
Supersonic		SUPR	Slot	SLOT
Internal		INTO	Injection	INJC
Mach		MACH	Coolant	COOA
2.4		2X25	Mass	MASZ
Non-adiabatic		ADIA	Flow	FLOW
Wall		WALL	Ratio	RATI
Axial		AXIA	Variation	VART
Symmetry		SYMM	Steady-state	STBI
			Subsonic	SUBS
			Turbulent	TUBU
			Pipe	PIPE
			Equation	EQUA
			Prediction	PRED
			Heat transfer	HEAF
			Rates	RATE
			Estimation	ESTM
			Comparison	CMRI
			Experiment	EXPE
			Theory	THEY
			Instrumentation	INSM

Naval Ordnance Laboratory, White Oak, Md.  
(NOL technical report 64-65)  
AIR-FILM COOLING OF A SUPERSONIC NOZZLE (U)  
by Bing H. Lieu. 18 Aug. 1964. 13p. diagrs.,  
charts. (Aerodynamics research report 224)  
BuWeps task PR-10.

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This report presents a comparison between experimental data and theoretical calculations of air-film cooling of an axisymmetric, Mach 2.4 nozzle. The turbulent pipe flow equation of Dittus and Boelter was found to be applicable in predicting the heat transfer rates in the absence of film cooling. A modified version of the semi-empirical equation of Hatch and Papell was found applicable in estimating the film-cooled nozzle wall temperatures.

1. Wind tunnels -
2. Nozzles -
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4. Cooling
5. Film
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7. Title
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